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Rating Technique for Reciprocating Refrigeration Compressors

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RATING TECHNIQUE FOR RECIPROCATING REFRIGERATION COMPRESSORS

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ABSTRACT

As refrigeration compressor manufacturers introduce machines with improved efficiencies there is a need for updating performance data.

The compressor performance data which is published should be a true representation of the actual compressor performance, historically this has resulted in lengthy test programmes for the compressor manufacturer. A system is required which can accurately produce performance data from a manageable number of tests.

As a result of analysing the behaviour of reciprocating refrigeration compressors, relationships become apparent which could reasonably be expected to form the basis of a technique which will economically generate accurate performance data over the full matrix of operating conditions.

The basis of possible techniques will be described and justified by reference to the principles of operation of reciprocating compressors.

INTRODUCTION

Specifiers of refrigeration equipment are continually required to select compressors which best match the needs of a specific system. When considering packaged equipment, the incorrect selection of a compressor will result in additional development time and expense.

The existing techniques for providing performance data vary from one compressor manufacturer to another. Although tests may be carried out to the same procedure on the same machine by two test houses it is highly likely that the data sets generated will have significant differences. One cause of the differences is obviously test rig errors but a further potential difference would be the way in which the test results are linked together to provide published data.

When performance data is presented in graphical form it is not possible to test every possible point which can be read from the graph. If a number of tests have been carried out it is a very simple matter to fit smooth curves through the capacity and power test results and, providing there is only minimal difference between the test result and the fitted line, the test results and the fitted curve will normally be considered to be correct. Using this technique the data generated will be of acceptable accuracy if:-

a. There are no significant test errors.
b. The data is not extrapolated outside the test range.
c. Test points are sufficiently close to minimise interpolation errors.

FIGURE 1
FITTING ERROR (capacity) vs EVAPORATING TEMPERATURE
R22
Tc=45C

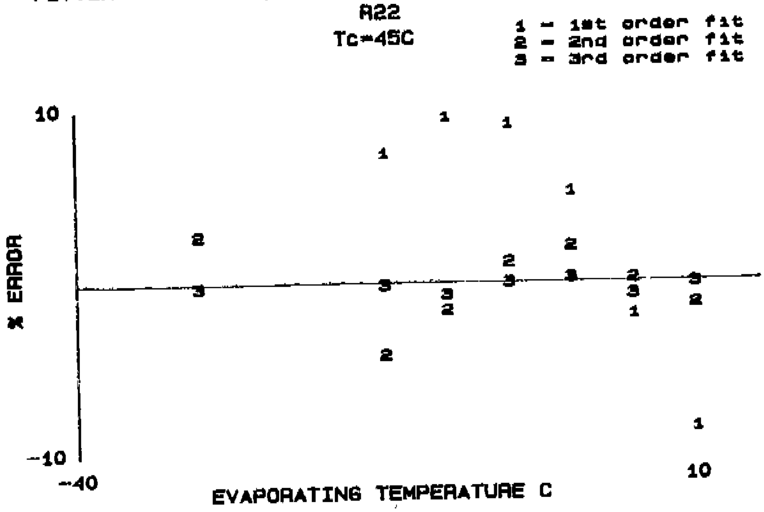
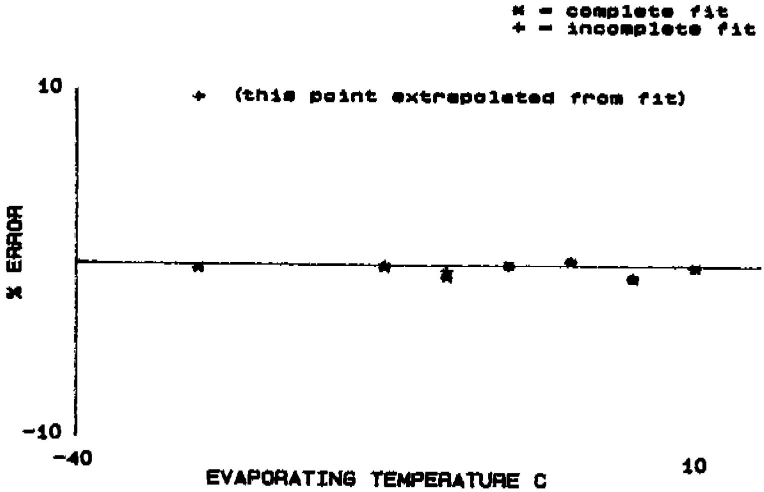


Figure 1 shows the effect of fitting capacity and power as a function of evaporating temperature using simple equations and it can be seen that a good fit is achieved by using a third order polynomial.

Figure 2 however indicates that the higher order polynomial, although providing a good fit within the band of accurate data, is unsuitable for extrapolating data outside the test range. This type of technique will also attempt to fit all data points, even when significant errors exist.

FIGURE 2
FITTING ERROR (capacity) vs EVAPORATING TEMPERATURE



The use of a second variable such as condensing temperature increases the risk of error when extrapolating, especially when the equations used to fit the capacity and power become more complex.

Even when a good fit to test data is achieved it is possible that consistent errors exist within the test data. A fitting technique is required which can fit lines of predetermined form to various aspects of the compressor's performance, these lines should be simple (preferably straight), as this most easily permits the identification of erroneous test points and allows a degree of extrapolation.

COMPRESSOR BEHAVIOUR

In order to provide performance data for a compressor, a technique is required which will allow the generation of capacity and motor or shaft power over a predetermined matrix of conditions. In the case of a motor compressor combination the current will also be required.

Capacity

The aspect of compressor performance most relevant to the equipment specifier is the refrigeration capacity, unfortunately this does not lend itself to a simple fitting technique.

A commonly recognised relationship is that, for a given condensing pressure, a straight line may reasonably be drawn when plotting volumetric efficiency against pressure ratio. In order to determine the validity of using a straight line fit various aspects affecting gas flow rates are to be considered.

In an ideal case the volumetric efficiency of a reciprocating compressor is given:-

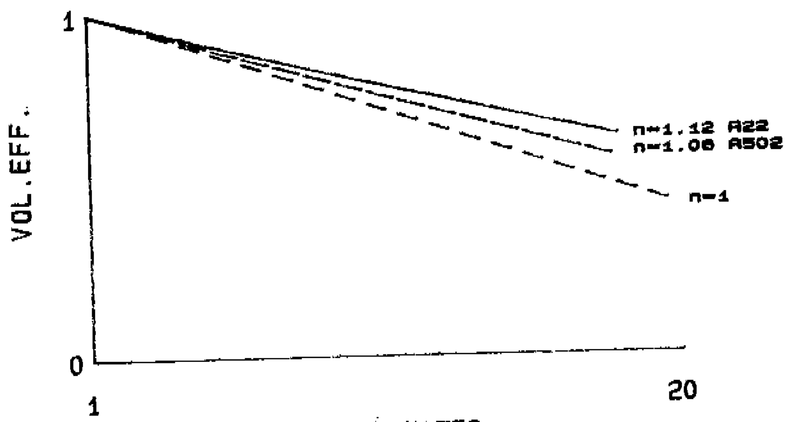
$$Veff = 1-Vc/Vs*(PR^{(1/n)}-1) \qquad [1]$$

where
- Veff = volumetric efficiency.
- Vc = compressor clearance volume m^3
- Vs = compressor displacement m^3
- PR = pressure ratio across compressor
- n = index of compression.

Inspection of equation 1 shows that a straight line relationship exists between volumetric efficiency and pressure ratio when the index of compression is unity. In practice the index for any of the commonly used fluorocarbon refrigerants does approach unity, giving only a slight deviation from a straight line fit, Figure 3.

FIGURE 3

IDEAL VOLUMETRIC EFFICIENCY Vc/Vs = 0.03



If the behaviour of reciprocating compressors was ideal then no further anlaysis would be needed. Few compressors are ideal and so consideration should be given to factors such as the following:-

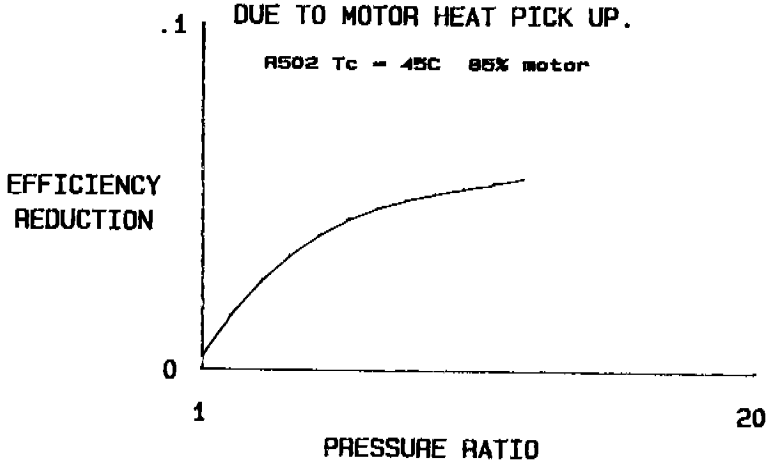
- Heat transfer to suction vapour within compressor.
- Valve timing.
- Internal leaks (piston blowby).
- Gas passage and valve pressure drops.

The effect of heat transfer to the suction vapour is to reduce the density of the gas entering the cylinder and thus reduce the mass of gas that is pumped by the compressor. The net result of the heat transfer is that the volume of gas pumped by the cylinders remains unaltered, but the volume flow corrected to the temperature at the service valve is reduced. Using the simplified assumption, for a suction gas cooled compressor, that the heat transfered to the suction vapour is the heat generated by the motor losses, then the reduction in flow would be as shown in Figure 4.

FIGURE 4

VOLUMETRIC EFFICIENCY REDUCTION

DUE TO MOTOR HEAT PICK UP.



Late closing of the automatic valves within a compressor also reduces capacity. The late closure of valves may result from the use of undersized porting, incorrect valve stiffness, or valve stiction. It is not possible to generalise on the exact behaviour of valves, but in most cases it would not be too unreasonable to consider that over a limited range of operating conditions the valve timing is constant and that at the instant of closure the pressure ratio across the valve is independent of operating condition. Using these simplified assumptions the reduction in flow due to late closing of the suction valve would be a constant, while late closing of the discharge valve would simulate additional clearance volume; Figure 5.

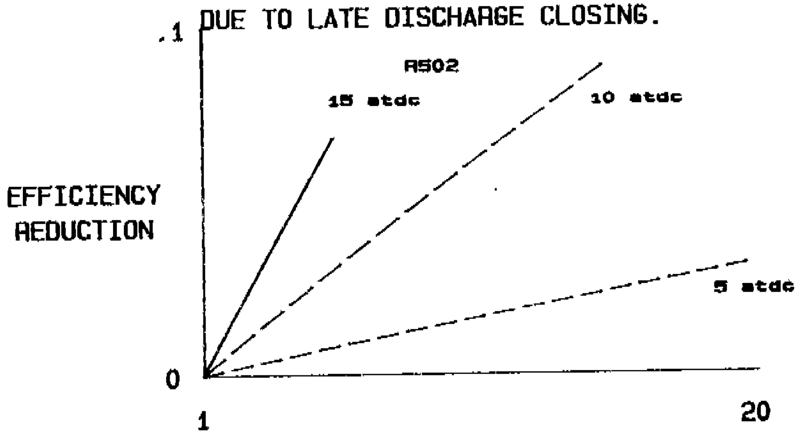
Internal leaks and piston blowby are difficult to predict and because of the small magnitude (typically less than 3% of flow) these are to be ignored within this analysis.

The effect on flow of pressure drops through the automatic valves will only be significant when these are sufficient to cause late closing of the valves. The effect of pressure drops through the gas passages can be more significant. Pressure drops through the discharge passages result in the pressure ratio across the cylinder being increased and thus the flow reduces as shown in equation 1. Pressure drops through the suction passages also increase the pressure ratio but in addition cause a density reduction between the service valve and the cylinder.

FIGURE 5

VOLUMETRIC EFFICIENCY REDUCTION

DUE TO LATE DISCHARGE CLOSING.



When combining all the flow losses, Figure 6, it can be seen that although a straight line is not the perfect solution when fitting volumetric efficiency against pressure ratio, it is certainly an option that offers only minimal errors. If the volumetric efficiency is based on a nominal displacement the reduction in motor speed at the low pressure ratios (high suction pressure) will tend to flatten the curve.

It can be demonstrated that, in most cases, the overall performance of test compressors exhibit a volumetric efficiency vs pressure ratio relationship, which for one condensing pressure tends, towards a straight line.

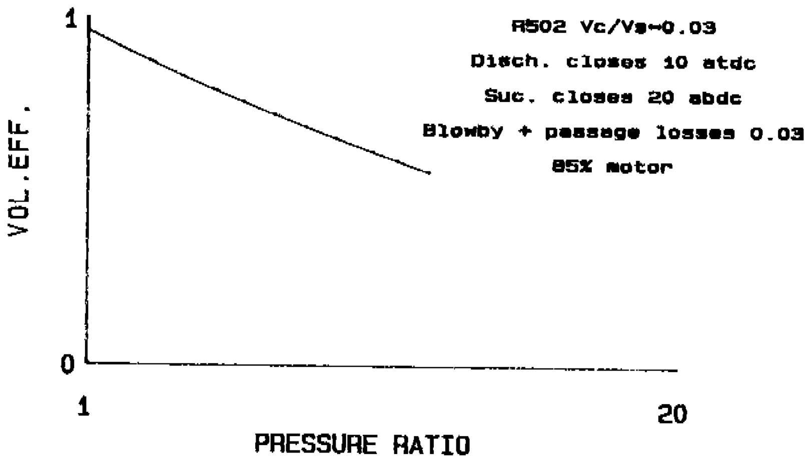
Power Input

The power curves for a reciprocating compressor are less symetrical than the capacity curves and are therefore even less suitable for fitting directly.

In trying to determine a suitable fit for describing the power input to a compressor, isentropic efficiency was considered. When plotting isentropic efficiency against pressure ratio a family of similar curve shapes is observed, Figure 7; but unfortunately not straight lines. By splitting the isentropic efficiency line at the maxima it is possible to define simple curve shapes which give an adequate fit. However, this technique relies on the extrapolation of volumetric efficiency data well outside the test range and an alternative method was therefore desirable.

FIGURE 6

THEORETICAL VOLUMETRIC EFFICIENCY



Analysis of isentropic efficiency provides several useful checks on performance data:-

Isentropic efficiency is always less than unity.

At a pressure ratio of 1 the isentropic efficiency is 0.

Projecting from the volumetric efficiency fit, when flow is 0 isentropic efficiency should be 0.

In considering how to determine the most suitable method for generating compressor power curves it is helpful to understand how this is made up. The major component (typically 55%-75% of motor input power) consists of the isentropic power.

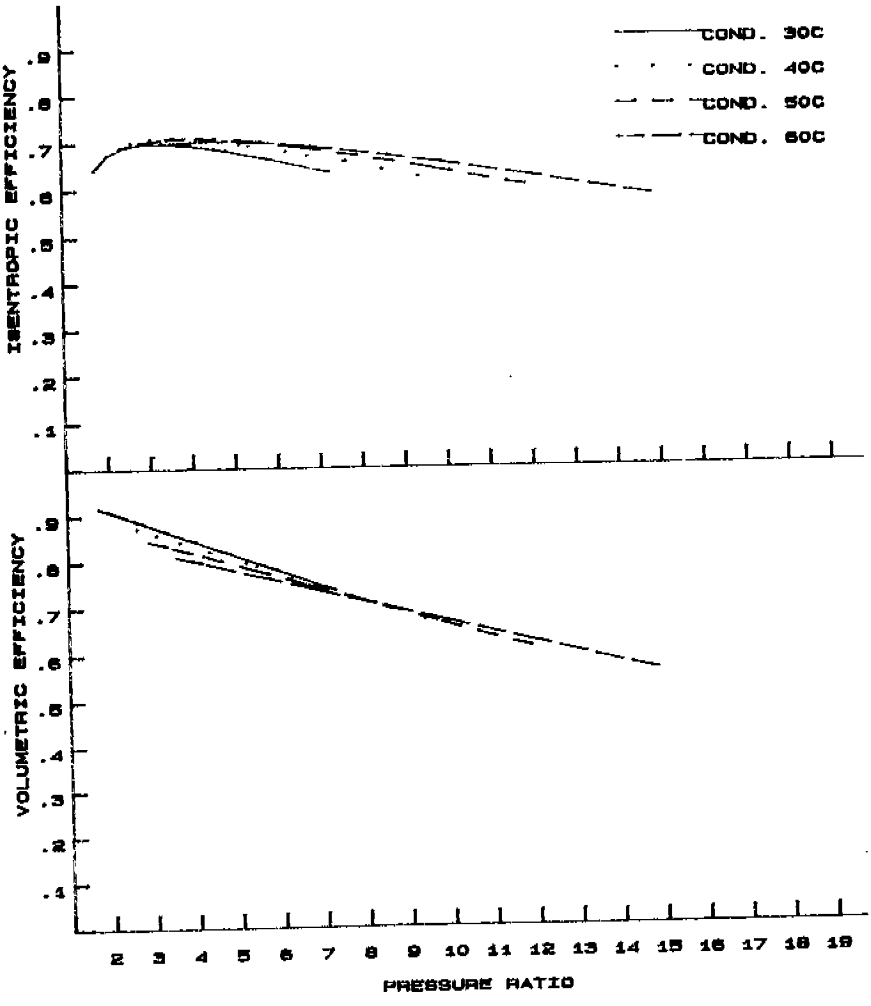
Isentropic work is defined:-

$$W = n/(n-1) * Ps * Vs * [PR^((n-1)/n)) -1] \qquad [2]$$

where

W	=	isentropic work done	W
Ps	=	suction pressure	N/m^2
Vs	=	induced volume of gas	m^3.

Figure 7. 3 cylinder. R22 28.4.86
FITTED EFFICIENCIES



In an ideal compressor the difference between the power input and the isentropic work would be zero, but in real machines the following will contribute to direct or indirect power losses:-

Direct

Friction/ancillary equipment losses.
Valve and gas passage pressure drops.
Motor losses (for semi hermetic or hermetic).

Indirect

Heat transfer.
Gas passage pressure drops.
Gas leaks.

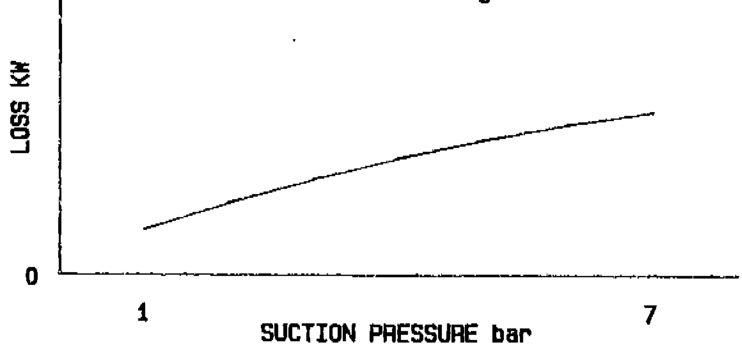
If power losses are only a minor part of the total input to a compressor and the isentropic power can be calculated directly from the refrigerant flow rate, then it is reasonable to assume that losses provide a basis for accurately fitting the compressor power.

When studying the compressor losses these must be expressed as a function of the operating condition and this is initially done with reference to suction pressure assuming that the discharge pressure remains constant.

Friction can be broken down into cylinder and bearing losses and in the case of an open compressor, shaft seal losses. An additional loss which can be considered as a frictional loss is the power required to drive the oil pump. The oil pump power can reasonably be considered to be constant over the full operating range of the compressor while bearing losses can be estimated using commercially available bearing data. An approximation can be made to the cylinder friction by integrating the piston and ring side loads and velocity and multiplying by a suitable friction coefficient, Figure 8.

FIGURE 8

CYLINDER FRICTION
6 cylinder 100m^3/hr R502
19 bar discharge



The effect of the pressure drop through the automatic (flapper) valves and gas passages can be simply calculated if it is assumed that the effective flow area of the valves and the gas velocity are constant during the cycle. It may be reasonable to assume that the gas velocity through the valves is constant as at high pressure ratios the flow is reduced but so is the time for which the valves are open. Calculating the valve losses in this way is only an approximation and the technique would be totally inadequate for calculating valve dynamics or gas flow rates.

Considering incompressible flow through the suction valve:-

$$PD = k * \rho * V^2 \qquad [3]$$

As velocity considered constant,

$$PD \propto \rho \qquad [4]$$

At a constant suction return temperature and with an index of compression that approaches unity it may be assumed:-

$$\rho \propto Ps \qquad [5]$$

and therefore

$$PD \propto Ps \qquad [6]$$

also

$$WD = PD * Vol \qquad [7]$$

and if a straight line fit of flow against pressure ratio is valid:-

$$Vol = k2 - k3 * Pd/Ps \qquad [8]$$

from equations [6] [7] & [8].

$$WD = k4*Ps - K5*Pd$$

or for a constant discharge pressure.

$$WD \propto Ps \qquad [10]$$

where

Pd	= discharge pressure	N/m^2
PD	= pressure drop	N/m^2
V	= velocity	m/s
Vol	= volume flow rate	m^3/s
WD	= work done/unit time	W
kn	= constant.	

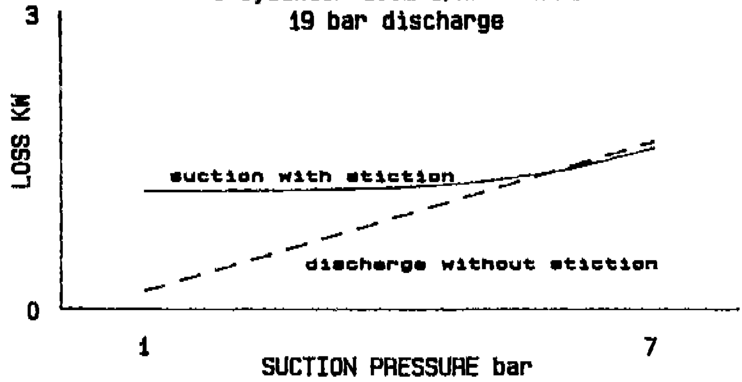
In practice the above relationship is affected by the valve reed sticking to the stationary plate, causing a significant pressure difference across the valve prior to opening and increased gas velocities due to reducing the time for which the valve is open. The result of stiction is therefore to produce valve losses which are no longer directly proportional to the gas pressure, Figure 9.

FIGURE 9

VALVE LOSSES

6 cylinder 100m^3/hr R502

19 bar discharge



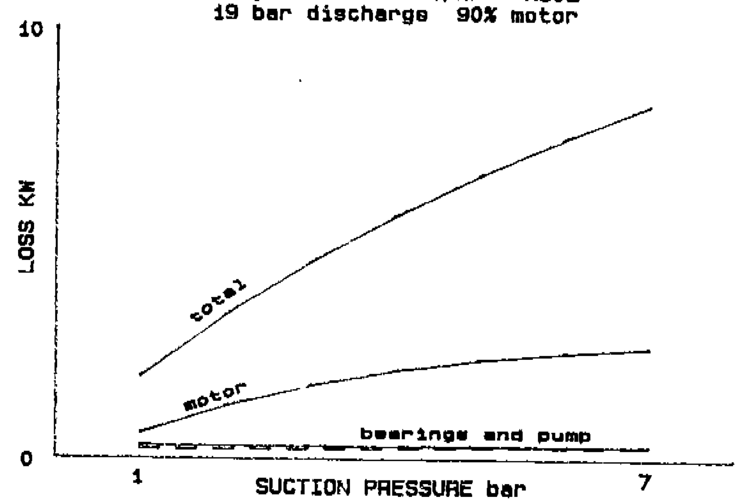
The motor used with semi hermetic refrigeration compressors have efficiencies typically between 85% and 90%. The motor loss will obviously vary with suction and discharge pressures, but the exact relationship between the loss and suction pressure will depend on the characteristics of a given motor and on the magnitude of other losses.

FIGURE 10

DIRECT LOSSES (no stiction)

6 cylinder 100m^3/hr R502

19 bar discharge 90% motor



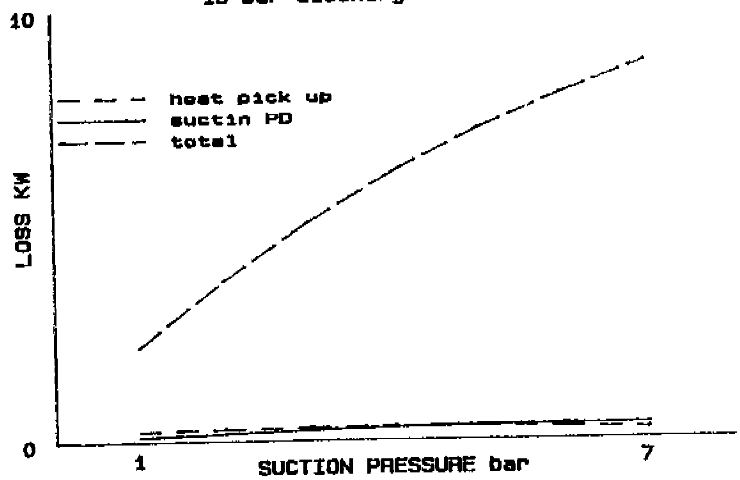
The direct power losses for a hypothetical compressor are shown in Figure 10. It can be seen that the fit of total direct losses against evaporating pressure approaches a straight line in this case.

In addition to losses which directly increase the power requirement there are certain factors which while not necessarily increasing the power input reduce the volume flow rate at the suction service valve. These losses, by reducing the volume flow, are increasing the power input per unit mass flow rate and can therefore be described as indirect power losses.

The first of the indirect losses is heat transfer to the suction vapour, this could occur in several ways but the example given in Figure 11 is based on this heat transfer being a single source; the motor losses. This loss is produced as previously described by increasing the specific volume of the refrigerant as it passes through the compressor.

FIGURE 11

TOTAL AND INDIRECT LOSSES
6 cylinder 100m^3/hr R502
19 bar discharge 90% motor



Pressure drops in the suction passages of the compressor reduce the volume flow by increasing the specific volume and increasing the pressure ratio at the cylinders. Indirect discharge passage losses result only from an increase in pressure ratio and therefore tend to be less significant than indirect suction losses.

Figure 11 shows the summation of the losses within the compressor, it can be seen that, although not a perfect straight line fit when plotted against evaporating pressure, the error introduced by using a straight line relationship will be small as the losses are a minor part of the total power input.

FIRST MODEL

It has been demonstrated that, for some compressors, straight lines can reasonably be used to fit volumetric efficiency against pressure ratio and power loss against evaporating pressure for a single condensing pressure. When test data is examined in some detail it becomes apparent that a different line exists for each condensing temperature and that the family of lines are not necessarily parallel.

A simple model which would appear to form the basis of a consistant fitting technique would be to define the volumetric efficiency and power losses as follows:-

$$\text{Volumetric efficiency} = C0 + C1 * PR + C2 * PR * Pd + C3 * Pd \qquad [11]$$

$$\text{Power loss} = C4 + C5 * Ps + C6 * Ps * Pd + C7 * Pd \qquad [12]$$

where:-

Cn = constant

isentropic power is calculated from fitted flow.

Using these relationships typically gives RMS errors in the order of 1% between test and fitted values, Table 1.

DEVIATION FROM FIRST MODEL

Not all compressors can be fitted to the same accuracy using this technique as some exist where the model described may not be the most appropriate.

In considering the power losses it was stated that the relationship between the losses and the suction presssure could vary with valve stiction and motor characteristics, this results in a better fit sometimes being obtained against evaporating temperature rather than pressure. If required it is a very simple matter for a fitting technique to compare two methods and select the most appropriate, as was done in the example given in Table 1.

Volumetric efficiency is not always a good straight line fit against pressure ratio but can sometimes tend more to an "S". This is an area where further studies are required to determine whether it is still acceptable to use straight lines.

MOTOR CURRENT

Motor currents should be very simple to fit but in many respects prove to be the most difficult if a straight line relationship is required.

A possible solution to finding a straight line fit is to fit the inductive current against power input. This technique gives good accuracy, as shown in Table 1, and is ideally suited when capacitors are required for power factor correction.

IN CONCLUSION

Using the basic technique as described it has been possible to improve the accuracy with which compressor performance data is generated, while at the same time offering the opportunity of shorter test programmes.

The adoption of a standard fitting technique of this type throughout the refrigeration industry could ultimately be of advantage to both compressor manufacturers and equipment specifiers.

TABLE 1

THIS DATA IS FITTED TO THE TEST POINTS CONTAINED IN FILE 'SAMPLE1'. 29.4.86
using fitting routine CRATE

REFRIGERANT R502 No. POINTS FITTED 16 380 VOLTS.

Ps	Pd	VOL EFF TEST	VOL EFF FIT	KW TEST	KW FIT	CURRENT TEST	CURRENT FIT	% ERROR FLOW	% ERROR POWER	% ERROR POWER	% ERROR AMPS
6.03	13.13	0.95	0.95	9.57	9.56	18.58	18.55	-0.12	-0.13	-0.15	-0.17
4.89	13.24	0.93	0.93	9.46	9.50	18.42	18.48	0.03	0.45	-0.03	0.32
3.48	13.18	0.90	0.91	8.67	8.71	17.23	17.23	0.73	0.38	0.03	-0.01
2.41	13.19	0.87	0.87	7.47	7.45	15.47	15.27	0.16	-0.33	0.60	-1.27
2.91	16.73	0.85	0.85	8.95	8.93	17.64	17.58	0.02	-0.25	0.05	-0.36
4.14	16.78	0.89	0.89	10.45	10.46	19.95	20.04	-0.03	0.11	-0.38	0.44
4.89	16.80	0.90	0.90	11.04	11.06	20.88	20.97	-0.01	0.23	-0.24	0.41
7.09	16.72	0.93	0.92	11.79	11.75	22.06	22.03	-0.38	-0.28	0.26	-0.15
7.09	21.01	0.89	0.89	13.98	13.95	25.68	25.58	-0.11	-0.22	0.19	-0.39
4.89	20.98	0.87	0.87	12.44	12.46	23.11	23.16	-0.09	0.17	-0.30	0.23
4.14	20.99	0.85	0.85	11.61	11.61	21.78	21.85	-0.14	0.01	-0.43	0.30
2.91	21.05	0.82	0.82	9.71	9.67	18.70	18.75	-0.77	-0.41	0.06	0.26
2.91	25.95	0.78	0.78	10.35	10.34	19.76	19.81	0.02	-0.05	0.62	0.26
4.14	25.63	0.82	0.82	12.59	12.62	23.30	23.40	0.36	0.24	-0.18	0.42
5.73	25.59	0.84	0.85	14.61	14.63	26.69	26.64	0.64	0.15	-0.16	-0.19
7.09	20.96	0.89	0.89	13.94	13.93	25.59	25.51	-0.29	-0.13	0.28	-0.32
**************************							RMS	0.350	0.253 (--*--)	0.305	0.434

SUCTION RETURN 20.0 °C 0.00 °C LIQUID SUBCOOLING. R502

DUTY (KW) EVAPORATING @ (°C)

COND. TEMP °C	-40	-35	-30	-25	-20	-15	-10	-5	0	5	10
30			14.32	18.07	22.45	27.54	33.43	40.21	47.99	56.89	67.09
40			12.24	15.54	19.39	23.85	29.01	34.94	41.72	49.48	58.33
50			10.31	13.15	16.45	20.29	24.70	29.76	35.55	42.13	49.61
60			8.55	10.92	13.68	16.86	20.52	24.71	29.47	34.86	40.96

POWER (KW) EVAPORATING @ (°C)

COND. TEMP °C	-40	-35	-30	-25	-20	-15	-10	-5	0	5	10
30			6.70	7.45	8.12	8.71	9.17	9.48	9.60	9.50	9.15
40			7.14	8.06	8.94	9.75	10.46	11.05	11.49	11.74	11.77
50			7.52	8.61	9.67	10.68	11.62	12.47	13.19	13.77	14.16
60			7.91	9.13	10.35	11.54	12.69	13.77	14.75	15.62	16.33

THIS DATA IS FITTED TO THE TEST POINTS CONTAINED IN FILE 'SAMPLE1'. 29.4.86
using fitting routine CRATE

REFRIGERANT R502 No. POINTS FITTED 5 380 VOLTS.

Ps	Pd	VOL EFF TEST	VOL EFF FIT	KW TEST	KW FIT	CURRENT TEST	CURRENT FIT	% ERROR FLOW	% ERROR POWER	% ERROR POWER	% ERROR AMPS
6.03	13.13	0.95	0.95	9.57	9.57	18.58	18.61	0.00	-0.01	0.02	0.15
2.41	13.19	0.87	0.87	7.47	7.47	15.47	15.38	-0.01	-0.05	0.14	-0.58
4.89	20.98	0.87	0.87	12.44	12.47	23.11	23.18	0.06	0.22	-0.47	0.30
2.91	25.95	0.78	0.78	10.35	10.34	19.76	19.84	-0.01	-0.07	0.18	0.40
7.09	20.96	0.89	0.89	13.94	13.93	25.59	25.50	-0.04	-0.11	0.20	-0.35
**************************							RMS	0.033	0.119 (--*--)	0.250	0.383

SUCTION RETURN 20.0 °C 0.00 °C LIQUID SUBCOOLING. R502

DUTY (KW) EVAPORATING @ (°C)

COND. TEMP °C	-40	-35	-30	-25	-20	-15	-10	-5	0	5	10
30			14.28	18.04	22.43	27.54	33.45	40.24	48.04	56.98	67.20
40			12.20	15.51	19.37	23.85	29.03	34.97	41.78	49.56	58.44
50			10.27	13.12	16.44	20.29	24.72	29.81	35.61	42.23	49.74
60			8.52	10.90	13.67	16.87	20.55	24.76	29.54	34.97	41.10

POWER (KW) EVAPORATING @ (°C)

COND. TEMP °C	-40	-35	-30	-25	-20	-15	-10	-5	0	5	10
30			6.72	7.47	8.14	8.73	9.19	9.49	9.61	9.51	9.15
40			7.15	8.07	8.95	9.76	10.47	11.06	11.50	11.75	11.78
50			7.52	8.61	9.67	10.68	11.63	12.48	13.20	13.77	14.16
60			7.90	9.13	10.35	11.54	12.69	13.77	14.76	15.62	16.33

ENERGETIC AND VOLUMETRIC CHARACTERISTICS FOR THE UNIFORM VALUATION OF GAS AND REFRIGERATION COMPRESSORS

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ABSTRACT

The uniform valuation of refrigeration and gas compressors is a basic supposition for the purposeful further development. The historically originated energetic and volumetric characteristics are not uniform despite of many efforts and in parts not clear in the signification. An analysis has been accomplished at the basis of the international literature as well as of rules and standards. A proposition has been explained under the consideration of the constructions enclosed to the motor with the object to unify the characteristics.

1. INTRODUCTION

Studying special literature and in cooperation with design and test engineers,in the teaching process as well as working as authors and editors of relevant special books at the same time we found out that is necessary to consider such fundamental questions of this subject which are assumed to have been solved for a long time. The uniform valuation of machines is a basic supposition for our engineers work,but it is not given in general for gas and refrigeration compressors.

First of all we want to consider the design and the development work and their valuation. For that reason great importance is due to the ideal machine as an orientation to the object aspired to. Often quite general principles are lost out of the field of view and that isn't well. The isothermic power input P_T and the isentropic power input P_S are the comparative